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Research of energetic balance of the hydraulic system with fixed displacement pump and pressure relief valve

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Abstract. *It was analyzed the basic schematics implementing energy level of industrial hydraulic systems. It was considered the structure and composition of the hardware system of fixed displacement pump and pressure relief valve with manual control. It was carried out a study of the energy balance of the hydraulic system of fixed displacement pump and pressure relief valve at 5 possible positions opening cross-section of adjustable throttle valve. For each of the provisions specified size and power consumption efficiency, as well as the value of energy loss of the hydraulic system. It is established distribution of energy losses between the actuator, pump, throttle valve and pressure relief valve under different operating conditions selected hydraulic system. The change in energy efficiency and the efficiency of the hydraulic system was determined under different operating conditions. Graphs change energy balance was built throughout the range of adjustment of the hydraulic system of fixed displacement pump and pressure relief valve.*

Keywords: *hydraulic system; energy balance; fixed displacement pump; pressure relief valve.*

Preface

Analyzing the hydraulic circuit of industrial systems should be noted that the main difference is their implementation of the power supply system, i.e. pump station [1-5]. The executive part is the same and the choice of actuators depends solely on operational and technological characteristics of each of the operations and in the design of the hydraulic system can not be changed by the developer. In fact, chosen of designer working pressure of system depends on selecting type and characteristics of the drive. Actually the choice of operating pressure depends on the designer, as there are levels of working pressures that are recommended for specific systems according to their capacity and that are reasonable in terms of economic and energetic feasibility and defined by size of hydraulic equipment. In practice, there are three levels of hydraulic system operating pressure, hydraulic systems with low (working pressure up to 15 MPa), medium (working pressure up to 25 MPa) and high pressure (working pressure above 25 MPa).

Baseline characteristics of pump station are the pressure and flow rate of fluid, which determine the useful power of the pump. Actually circuit implementation can be divided into groups depending on the possibilities of regulating the output characteristics of the pump. Thus, in general, all possible circuit implementation pumping stations can be divided in 4 groups [6-8]:

1. Systems with constant pressure and constant flow rate;
2. Systems with variable pressure and constant flow rate;
3. Systems with constant pressure and variable flow rate;
4. Systems with variable pressure and variable flow rate.

Note that this dividing of system on consumption is legitimate, as the flow created by the pump depends exclusively on the characteristics of the pump (working displacement of pump and velocity of the drive shaft) and therefore it can really be fixed or variable, but situation with working pressure is not so straightforward. As is known, the pressure in the system is not determined by the pump, but by the load, which creates a hydraulic resistance of system, so to speak about pumping stations with constant or variable pressure, is wrong. Therefore by proposed classification we are referring not to the current working pressure in the system, but to the maximum possible pressure. So we get two types of systems with constant maximum pressure and possible maximum pressure that can vary during the system operation automatically without human intervention. Although it is understood that in the system with a constant maximum pressure, working pressure at a time of the system may differ from the maximum and even be

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almost zero (if the actuators don't have load), i.e. not be constant. This fact obviously complicates the understanding of energy processes that occur in the system and requires the previous simulation.

To the first group (of constant pressure and flow) we refer hydraulic systems with fixed displacement pump and pressure relief valve.

To the second group (systems with variable pressure and constant flow rate) we refer systems with variable displacement pump and: a) differential pressure valve and pressure relief valve; b) proportional pressure relief valve;

To the third group (systems with constant pressure and variable flow rate) we refer the systems with: a) pumps with adjustable pressure regulator b) two- and many pumps system;

To the fourth group (systems with variable pressure and flow) we refer the systems with: a) variable displacement pumps with LS (Load-Sensing) regulator; b) LUDV (High Performance Flow Sharing) control; c) hydraulic accumulator.

Consideration of proposed systems will be produced with an emphasis on determining the level of efficiency, consumed energy and sizes, locations and causes of energy losses.

Presentation of main material

In this article we will investigate the energy balance of hydraulic systems on the example of fixed displacement pump and pressure relief valve.

This type of system is one of the easiest and cheapest by implementation, because the energy level is composed of fixed displacement pump and pressure relief valve with manual setting of maximum pressure in the system (Fig. 1).

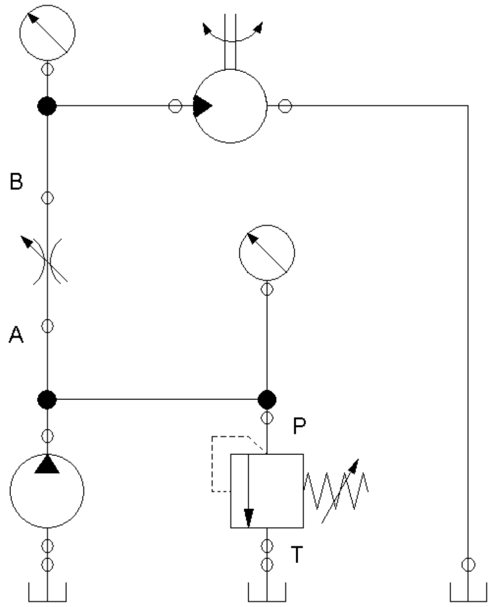

Fig. 1. Hydraulic system with fixed displacement pump and pressure relief valve

For a better understanding of energy processes which occur in such system we will use output characteristics of the system like nominal flow rate of pump – 60 l / min, operating opening pressure of pressure relief valve is 130 bar, the load on the shaft of the hydraulic motor corresponds to the pressure of 60 bar, full coefficient of efficiency of hydraulic pumps and motors is 0.9.

We consider the energy balance of the system with 5 possible positions of throttle valve: 1 - completely open; 2 - open 75%; 3 - open 50%; 4 - open 25%; 5 - completely closed.

1. In the first variant, when the adjustable throttle valve is fully opened, because we neglect hydraulic losses along the length of pipelines and in local hydraulic resistances, this scheme can be compared to the same scheme without throttle valve, i.e. hydraulic fluid enters the hydraulic motor directly from the pump.

The pressure in the system in this variant will be determined by load on the hydraulic motor shaft and will be equal to 60 bars. Flow rate of the working fluid created by the pump is constant because the pump is with fixed displacement and has constant rotational speed of the drive shaft, and completely fed to the hydraulic motor. In such scheme useful power is spent actually identical except for losses, which arising in the pump, the motor and the throttle valve, and will be determined by pressure of load and flow rate of pump.

Even at full opened throttle valve, it occur hydraulic losses of the working fluid. For example, make use of the chart "Pressure drop – Flow rate" of industrial throttle valve FG16K70-2X / V from Bosch Rexroth (Fig. 2).

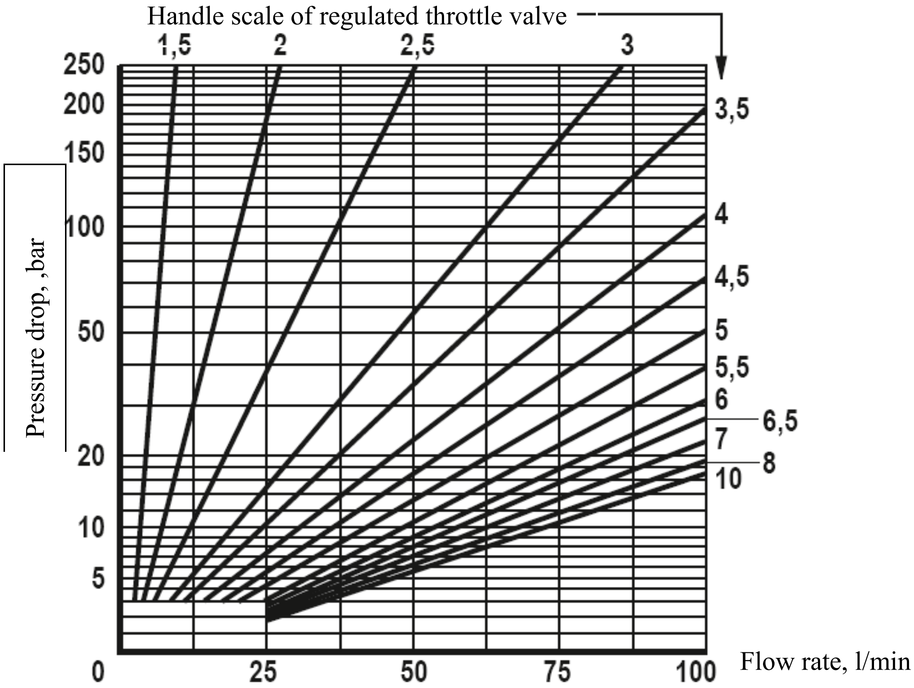

Fig. 2. Characteristic Δp - qv (Pressure drop – Flow rate) of regulated throttle valve NG16 type

As is shown in this diagram, when throttle valve is fully opened (mark on the throttle valve handle and curve 10 in diagram) by passing through the throttle valve flow rate 60 l/min occurs the pressure drop at 8 bars. Thus, the pressure before the throttle valve will consist in pressure of load on the drive and pressure drop on the throttle and equals 68 bars. As this pressure is lower than the opening pressure of pressure relief valve, all flow rate will perform to useful work of hydraulic motor (Fig. 3).

$$N_{USF} = P_{MOT} \cdot Q_{MOT} \cdot \eta_{MOT} = 6 \cdot 10^6 \cdot 1,0 \cdot 10^{-3} \cdot 0,9 = 5400W = 5,4kW ;$$

$$N_{SUP} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} = \frac{6,8 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} = 7556W = 7,6kW ;$$

$$N_{LOS} = N_{SUP} - N_{USF} = 7556 - 5400 = 2156W = 2,2kW .$$

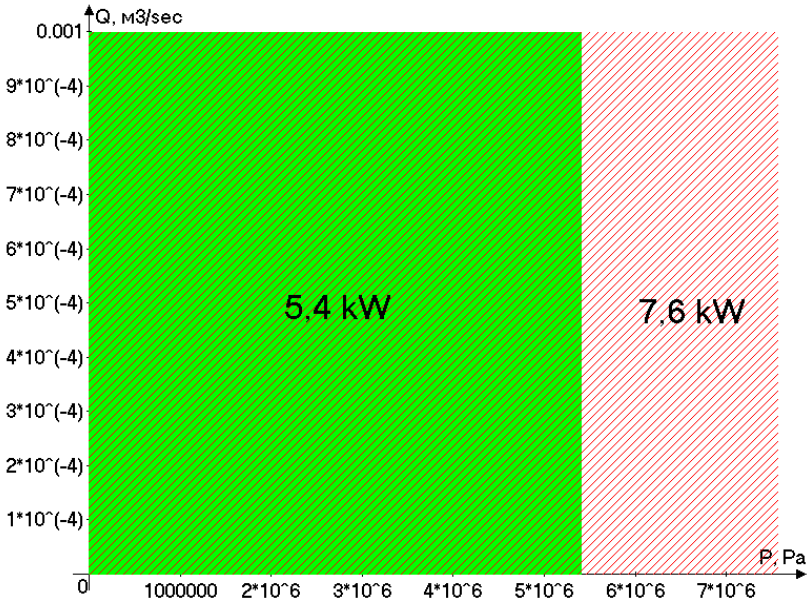

Fig. 3. Useful and supplied power in the first variant: **– useful power;** **– supplied power**

As is it seen from the calculation in this version, all the power consumed by the pump, excluding losses in the pump and motor, is used to perform useful work. The efficiency of the system for this mode without losses is equal 100%, and the actual efficiency with taking into account losses in the pump, motor and throttle valve is equal:

$$\eta_{SYS} = \frac{N_{USF}}{N_{SUP}} = \frac{5400}{7556} \cdot 100\% = 71\% .$$

Energy losses arising in this mode of system will be distributed between the pump, motor and throttle valve.

$$N_{LOS}^{PUM} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} \cdot (1-\eta_{PUM}) = \frac{6,8 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} \cdot 0,1 = 756W = 0,8kW ;$$

$$N_{LOS}^{MOT} = P_{MOT} \cdot Q_{MOT} \cdot (1-\eta_{MOT}) = 6 \cdot 10^6 \cdot 1,0 \cdot 10^{-3} \cdot 0,1 = 600W = 0,6kW ;$$

$$N_{LOS}^{THR} = (P_{PUM} - P_{MOT}) \cdot Q_{MOT} = 0,8 \cdot 10^6 \cdot 1,0 \cdot 10^{-3} = 800W = 0,8kW .$$

The energy balance of the system will be as follows (Fig. 4).

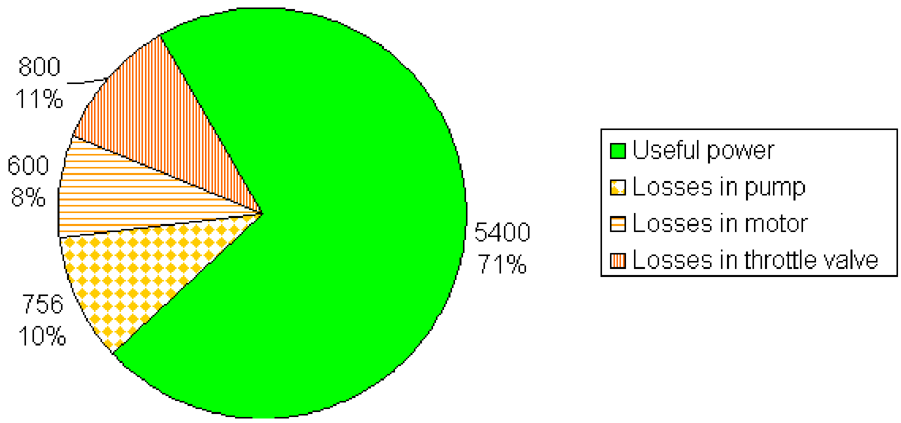

Fig. 4. Energy balance of power in first version of operation

2. The second variant considers the previous diagram (Fig. 1) with the opening of the throttle valve in cross-section, which corresponds to 75% of the flow rate of the pump (corresponding regulator throttles to 2.6 according to the throttle valve flow rate characteristic in Fig. 2). In this case, it means that 45 l / min will pass through the throttle valve and served actuator, and 15 L / min will merge through the pressure relief valve with pressure according to flow rate characteristics of pressure relief valve (Fig. 5), such as for example industrial pressure relief valve DB6DPW7-1X / 160V (Bosch Rexroth) at 142 atm.

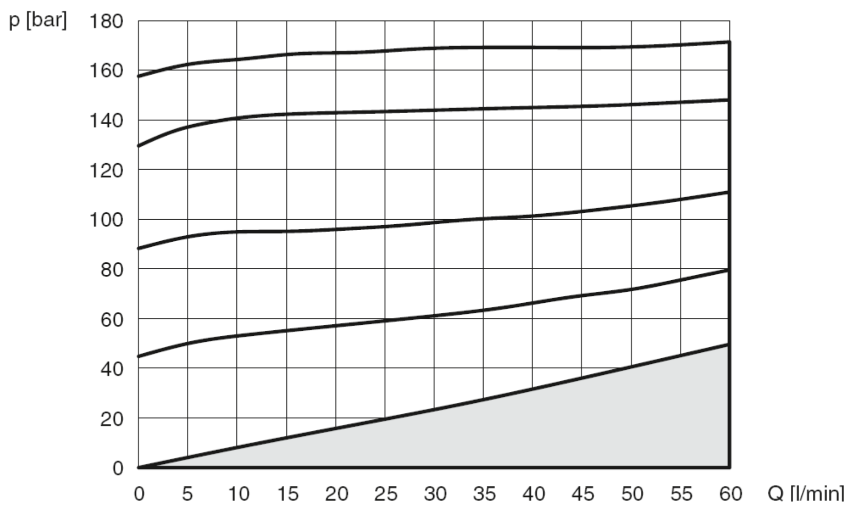

Fig. 5 Characteristic p - qv (Pressure – Flow rate) of pressure relief valve

Thus, due to regulating the flow of the working fluid supplied to the actuator was a change of useful and supplied power (Fig. 6).

$$N_{USF} = P_{MOT} \cdot Q_{MOT} \cdot \eta_{MOT} = 6 \cdot 10^6 \cdot 7,5 \cdot 10^{-4} \cdot 0,9 = 4050W = 4,1kW;$$

$$N_{SUP} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} = \frac{14,2 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} = 15778W = 15,8kW;$$

$$N_{LOS} = N_{SUP} - N_{USF} = 15778 - 4050 = 11728W = 11,7kW.$$

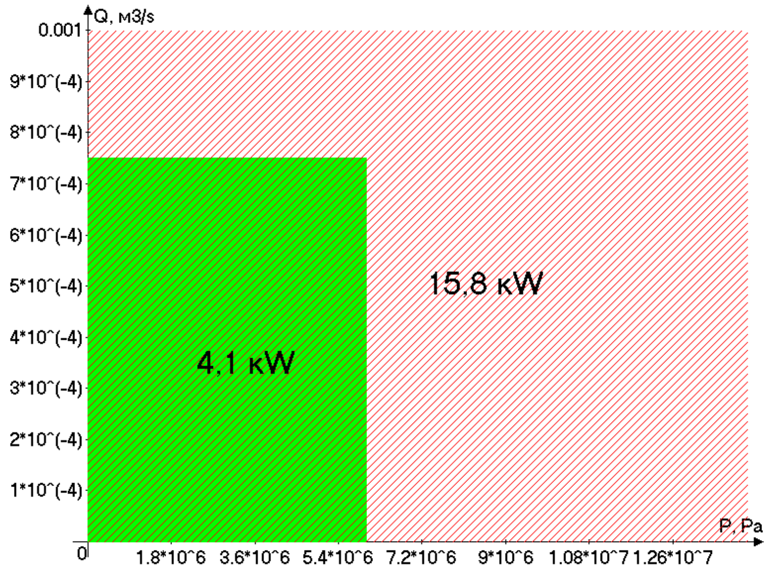

Fig. 6. Useful and supplied power in the second variant: – useful power; – supplied power

Accordingly, the efficiency of the same system in this version of the work will be reduced.

$$\eta_{SYS} = \frac{N_{USF}}{N_{SUP}} = \frac{4050}{15778} \cdot 100\% = 26\%.$$

In this case the energy losses that occur in this mode of system will be distributed not only between the pump, motor and throttle valve, but also pressure relief valve.

$$N_{LOS}^{PUM} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} \cdot (1 - \eta_{PUM}) = \frac{14,2 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} \cdot 0,1 = 1578W = 1,6kW;$$

$$N_{LOS}^{MOT} = P_{MOT} \cdot Q_{MOT} \cdot (1 - \eta_{MOT}) = 6 \cdot 10^6 \cdot 7,5 \cdot 10^{-4} \cdot 0,1 = 450W = 0,5kW;$$

$$N_{LOS}^{THR} = (P_{PUM} - P_{MOT}) \cdot Q_{MOT} = 8,2 \cdot 10^6 \cdot 7,5 \cdot 10^{-4} = 6150W = 6,2kW;$$

$$N_{LOS}^{PRV} = (P_{PUM} - P_{TAN}) \cdot Q_{TAN} = 14,2 \cdot 10^6 \cdot 2,5 \cdot 10^{-4} = 3550W = 3,6kW.$$

The energy balance of the system for the second variant will be as follows (Fig. 7).

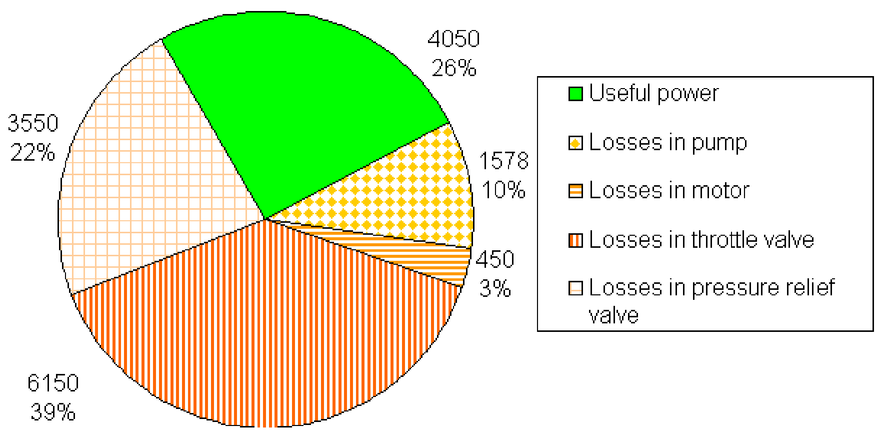

Fig. 7. Energy balance of power in second version of operation

3. In the third version we consider the previous diagram (Fig. 1) with the opening of the throttle valve in cross-section, which corresponds to 50% of flow rate of the pump (corresponding regulator throttles to 2.5 according to the throttle valve characteristic in Fig. 2). In this case, it means that 30 l/min will pass through the throttle valve and served actuator, and 30 l/min will flow through the pressure relief valve under pressure according pressure relief valve characteristics 145 bar (Fig. 5).

Thus, due to control the flow rate of the working fluid supplied to the actuator can be changed useful and supplied power (Fig. 8).

$$N_{USF} = P_{MOT} \cdot Q_{MOT} \cdot \eta_{MOT} = 6 \cdot 10^6 \cdot 5,0 \cdot 10^{-4} \cdot 0,9 = 2700W = 2,7kW\ ,$$

$$N_{SUP} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} = \frac{14,5 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} = 16111W = 16,1kW\ .$$

$$N_{LOS} = N_{SUP} - N_{USF} = 16111 - 2700 = 13411W = 13,4kW$$

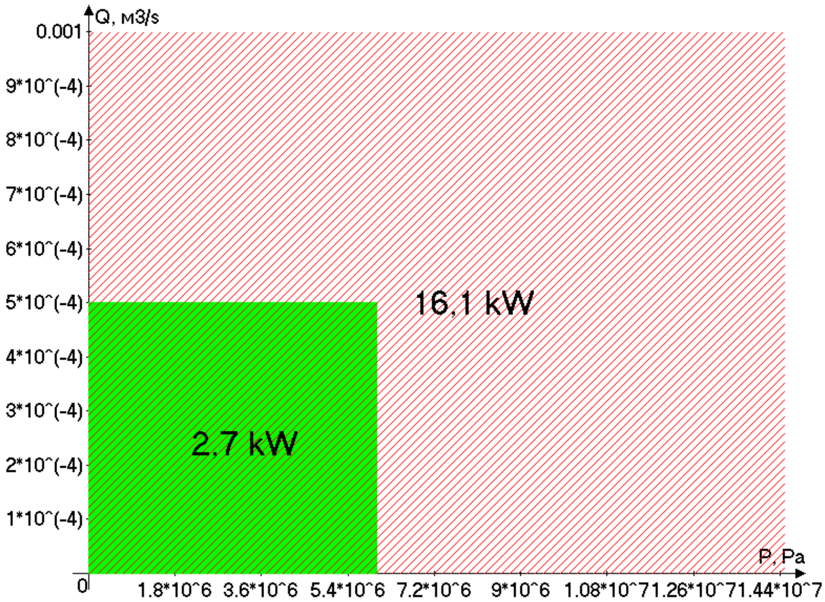

Fig. 8. Useful and supplied power in the third variant: – useful power; – supplied power

Accordingly, the efficiency of the same system in this version of the work will be reduced.

$$\eta_{SYS} = \frac{N_{USF}}{N_{SUP}} = \frac{2700}{16111} \cdot 100\% = 17\%\ .$$

In this case the energy losses that occur in this mode of system will be distributed as in previous variant between the pump, motor, throttle valve and pressure relief valve.

$$N_{LOS}^{PUM} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} \cdot \left(1 - \eta_{PUM}\right) = \frac{14,5 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} \cdot 0,1 = 1611W = 1,6kW$$

$$N_{LOS}^{MOT} = P_{MOT} \cdot Q_{MOT} \cdot \left(1 - \eta_{MOT}\right) = 6 \cdot 10^6 \cdot 5,0 \cdot 10^{-4} \cdot 0,1 = 300W = 0,3kW$$

$$N_{LOS}^{THR} = \left(P_{PUM} - P_{MOT}\right) \cdot Q_{MOT} = 8,5 \cdot 10^6 \cdot 5,0 \cdot 10^{-4} = 4250W = 4,2kW$$

$$N_{LOS}^{PRV} = \left(P_{PUM} - P_{TAN}\right) \cdot Q_{TAN} = 14,5 \cdot 10^6 \cdot 5,0 \cdot 10^{-4} = 7250W = 7,2kW$$

The energy balance of the system for the third variant will be as follows (Fig. 9).

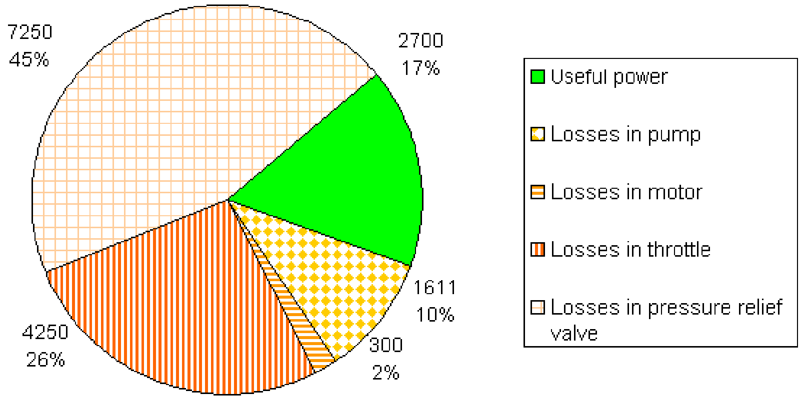

Fig. 9. Energy balance of power in third version of operation

4. In the fourth version we consider the previous diagram (Fig. 1) with the opening of the throttle valve in cross-section, which corresponds to 25% of flow rate of the pump (corresponding regulator throttles to 1.8 according to the characteristics of the throttle characteristic in Fig. 2). In this case, it means that 15 l/min will pass through the throttle valve and served actuator, and 45 l/min will flow through the pressure relief valve under pressure according pressure relief valve characteristics 148 bar (Fig. 5).

Thus, due to control the flow rate of the working fluid supplied to the actuator can be changed useful and supplied power (Fig. 10).

$$N_{USF} = P_{MOT} \cdot Q_{MOT} \cdot \eta_{MOT} = 6 \cdot 10^6 \cdot 2,5 \cdot 10^{-4} \cdot 0,9 = 1350W = 1,4kW \ ,$$

$$N_{SUP} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} = \frac{14,8 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} = 16444W = 16,4kW \ ;$$

$$N_{LOS} = N_{SUP} - N_{USF} = 16444 - 1350 = 15094W = 15,1kW \ .$$

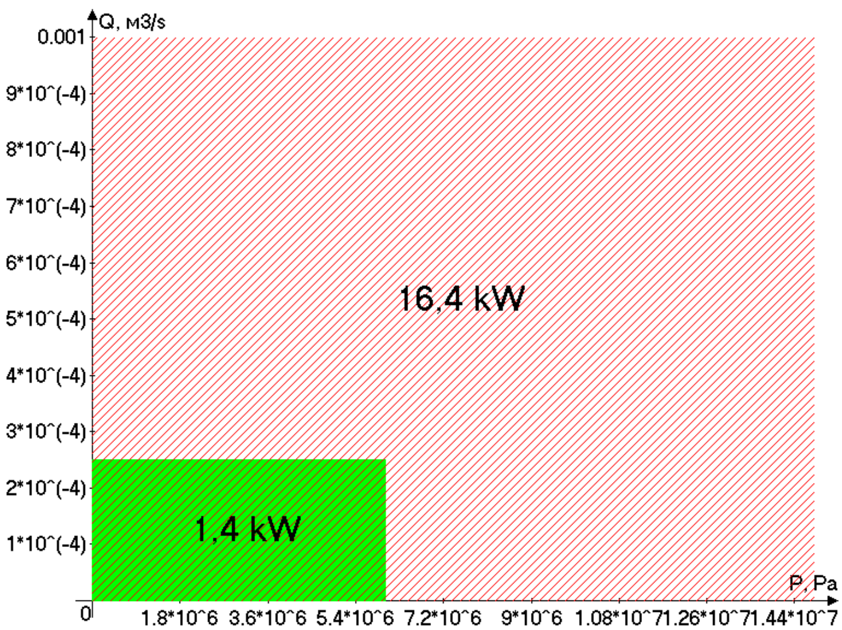

Fig. 10. Useful and supplied power in the fourth variant: – useful power; – supplied power

Accordingly, the efficiency of the same system in this version of the work will be reduced.

$$\eta_{SYS} = \frac{N_{USF}}{N_{SUP}} = \frac{1350}{16444} \cdot 100\% = 8\% .$$

In this case the energy losses that occur in this mode of system will be distributed as in previous variant between the pump, motor, throttle valve and pressure relief valve.

$$N_{LOS}^{PUM} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} \cdot (1-\eta_{PUM}) = \frac{14,8 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} \cdot 0,1 = 1644W = 1,6kW ;$$
$$N_{LOS}^{MOT} = P_{MOT} \cdot Q_{MOT} \cdot (1-\eta_{MOT}) = 6 \cdot 10^6 \cdot 2,5 \cdot 10^{-4} \cdot 0,1 = 150W = 0,2kW ;$$
$$N_{LOS}^{THR} = (P_{PUM} - P_{MOT}) \cdot Q_{MOT} = 8,8 \cdot 10^6 \cdot 2,5 \cdot 10^{-4} = 2200W = 2,2kW ;$$
$$N_{LOS}^{PRV} = (P_{PUM} - P_{TAN}) \cdot Q_{TAN} = 14,8 \cdot 10^6 \cdot 7,5 \cdot 10^{-4} = 11100W = 11,1kW .$$

The energy balance of the system for the fourth variant will be as follows (Fig. 11).

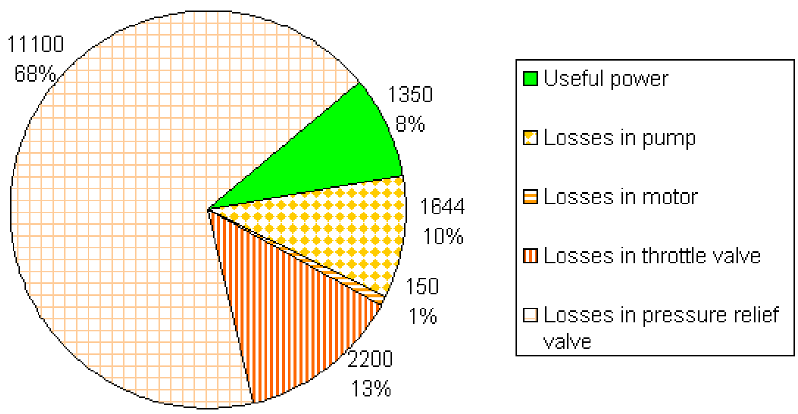

Fig. 11. Energy balance of power in fourth version of operation

5. In the fifth version we consider the previous diagram (Fig. 1) with fully closed throttle valve or such situation, when actuator reached end position or external load is higher than generated force. It means, that all flow rate of the pump will pass through the pressure relief valve with pressure 150 bar according to pressure relief valve characteristics (Fig. 5). All supply power will convert into losses in pump and pressure relief valve.

Hydraulic motor is stopped and useful power is zero, appropriately total efficiency will be zero also. Thus, due to control the flow rate of the working fluid supplied to the actuator can be changed useful and supplied power.

$$N_{SUP} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} = \frac{15 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} = 16667W = 16,7kW ;$$
$$N_{LOS}^{PUM} = \frac{P_{PUM} \cdot Q_{PUM}}{\eta_{PUM}} \cdot (1-\eta_{PUM}) = \frac{15 \cdot 10^6 \cdot 1,0 \cdot 10^{-3}}{0,9} \cdot 0,1 = 1667W = 1,7kW ;$$
$$N_{LOS}^{PRV} = (P_{PUM} - P_{TAN}) \cdot Q_{TAN} = 15 \cdot 10^6 \cdot 1 \cdot 10^{-3} = 15000W = 15kW .$$

The energy balance of the system for the fifth variant will be as follows (Fig. 12).

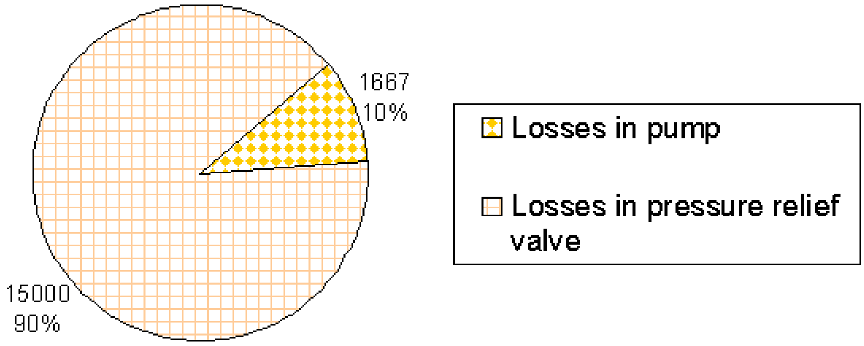

Fig. 12. Energy balance of power in fifth version of operation

The real distribution of energy losses, for example, for the third variant of the system's operation, has the following corelation (Fig. 13). At the same time, in order to simplify the calculations, we assume that the full efficiency of the pump and the motor ($\eta_{PUM} = \eta_{MOT} = 0,9$) is a product of two equal in size volumetric and hydromechanic efficiencies $\eta^{VOL}_{PUM} = \eta^{HYD}_{PUM} = 0,95$ and $\eta^{VOL}_{MOT} = \eta^{HYD}_{MOT} = 0,95$.

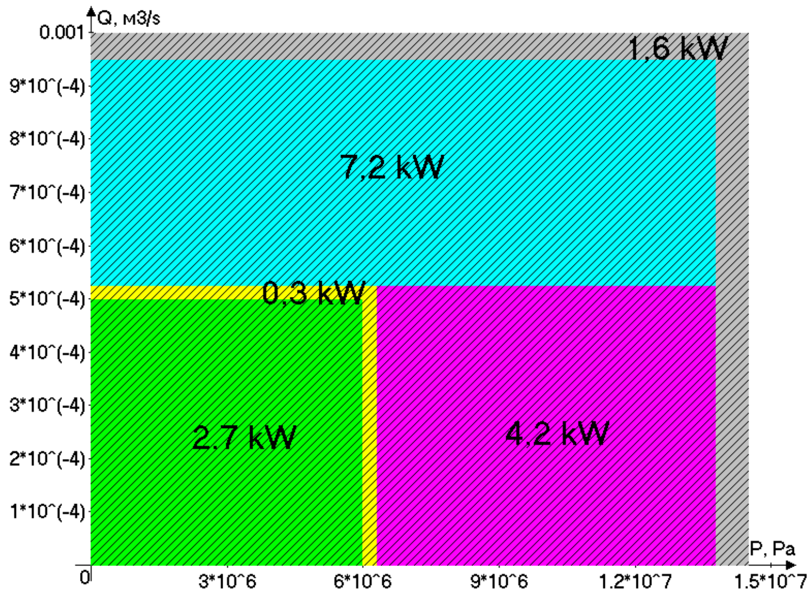

Fig. 13. Supplied and useful power in the third variant of operation: – supplied power (16 kW); – useful power (2,7 kW); – losses in hydraulic motor (0,3 kW); – losses in throttle valve (4,2 kW); – losses in pressure relief valve (7,2 kW); – losses in pump (1,6 kW)

Conducted studies have shown that even for such simple system with constant load on the actuator there is a significant redistribution of energy balance. Actually, the graphic of useful power is obvious, since when opening or closing the throttle valve defines a change in the flow of the working fluid, and as a consequence the change of speed. This change of the actuator speed causes a change of useful power, which is directly proportional to the size of the throttle valve opening.

Supplied power is also variable, but as can be seen from the study (Fig. 14), this change is also nonlinear. Actually, the curve of supplied power consists of three parts, in the first (from 10 to 3.6) and the third (from 2.5 to 0) of which there is a slow increase in power consumption, and in the second part (from 3.6 to 2.5) at the moment after the opening of the pressure relief valve occurs rapid increase of supplied power. The first part determines the change of power only by increasing the losses on the throttle valve when it is closing, and the second and third parts are determined by the total losses on the throttle valve and the pressure relief valve after it is opened.

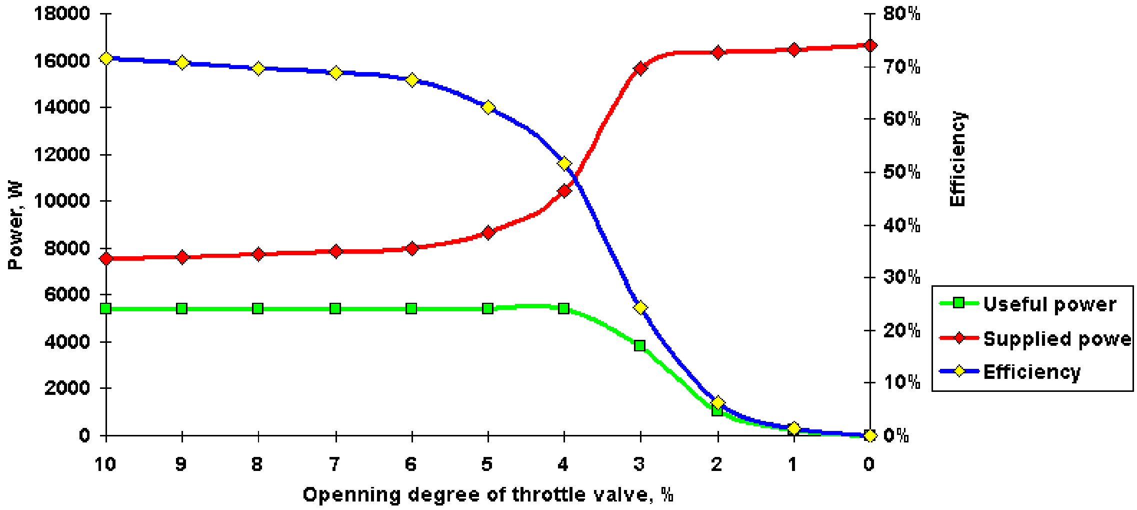

Fig. 14. Diagram of energy balance by different opening degree of throttle valve

For our example, the transition point from the first linear power consumption curve to the second according to the characteristics of the selected equipment (Fig. 2 and 5) is in the position 3.6 of the knob of the throttle valve regulator. That is, during the closing of the throttle valve (from position 10 to position 3.6 of the throttle valve knob) occurs an increase in power consumption by reducing the cross-section, but the overflow valve remains closed. This section is a kind of "dead zone" of the throttle valve, because energy consumption is increasing, but the expected result, the decrease of the speed of the actuator, doesn't occur. The speed doesn't increase because the pressure before throttle valve doesn't reach the pressure of opening the pressure relief valve 130 bar and the entire flow of the pump is fed through a smaller section of the throttle valve, but with a large pressure difference to the actuator.

Summary

The research of the hydraulic system with constant pump and pressure relief valve for the purpose of changing the level of energy consumption and, accordingly, the level of energy efficiency showed that the efficiency of such a system is not constant and depends not only on the efficiency of the hydraulic equipment, that are part of the system, but also on the operating modes of the system itself. Thus, the efficiency of the system is determined by two factors: the efficiency of the used equipment and the operating modes of the system. The first factor depends on the accuracy and quality of the manufacture of hydraulic devices and, during its operation, efficiency decreases as a result of wear and clearance increase of friction pairs. As a result, this change in the efficiency of the apparatus is almost impossible because it usually used standard industrial hydraulic equipment, where is very difficult to reduce wear. This wear is largely dependent on the duration of the operation of the hydraulic equipment. The second factor, as the research showed, varies depending on the circuit and operating modes of the system. Indeed, carrying out such studies with all known systems of a volume hydraulic drive will determine the tasks and areas of use in which standard hydraulic systems will be most effective.

Study of the energy balance of a system with fixed displacement pump and pressure relief valve was carried out with a change of opening value the throttle valve, but with a fixed conditional load. Therefore, the next direction of research will be systems with variable load on the actuator.

Дослідження енергетичного балансу гідравлічної системи з нерегульованим насосом і переливним клапаном

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Анотація. *Проаналізовано основні схемні реалізації енергетичного рівня промислових гідравлічних систем. Було розглянуто структуру та склад апаратної частини системи з нерегульованим насосом та переливним клапаном з ручним регулюванням. Проведено дослідження енергетичного балансу гідравлічної системи з нерегульованим насосом та переливним клапаном при 5-ти можливих поперечних перерізах регульованого дроселя. Для кожного з положень визначено величину та ефективність споживання енергії, а також величину втрат енергії гідравлічної системи. Встановлено розподіл втрат енергії між виконавчим пристроєм, насосом, дроселем та переливним клапаном при різних режимах* *роботи гідравлічної системи. Зміна енергоефективності гідравлічної системи визначалася при різних умовах експлуатації. Графіки зміни енергетичного балансу були отримані в межах діапазону регулювання гідравлічної системи з нерегульованим насосом та переливним клапаном.*

Ключові слова: *гідравлічна система; енергетичний баланс; нерегульований насос; переливний клапан.*

Исследование энергетического баланса гидравлической системы с нерегулируемым насосом и переливным клапаном

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Аннотация. *Проанализировано основные схемные реализации энергетического уровня промышленных гидравлических систем. Была рассмотрена структура и состав аппаратной части системы с нерегулируемым насосом и переливным клапаном с ручным регулированием. Проведено исследования энергетического баланса гидравлической системы с нерегулируемым насосом и переливным клапаном при 5-ти возможных поперечных сечениях регулируемого дросселя. Для каждого из положений определено величину и эффективность потребления энергии, а также величину потерь энергии гидравлической системы. Определено распределение потер энергии между исполнительным устройством, насосом, дросселем и переливным клапаном при разных режимах работы гидравлической системы. Изменение энергоэффективности гидравлической системы определялась при разных условиях эксплуатации. Графики изменения энергетического баланса были получены в границах диапазона регулирования гидравлической системы с нерегулируемым насосом и переливным клапаном.*

Ключевые слова: *гидравлическая система; энергетический баланс; нерегулируемый насос; переливной клапан.*

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